NASA TECHNICAL NOTE

NASA TN D-6351

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NASA TN D-6351

OPTIMIZATION OF SELF-ACTING HERRINGBONE JOURNAL BEARINGS FOR MAXIMUM RADIAL LOAD CAPACITY

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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • MAY 1971

1. Report No. NASA TN D-6351	2. Government Accession No.	3. Recipien.
4. Title and Subtitle OPTIMIZATION OF SELF-ACTING HERRINGBONE JOURNAL BEARINGS FOR MAXIMUM RADIAL LOAD CAPACITY		5. Report Date May 1971
		6. Performing Organization Code
7. Author(s) Bernard J. Hamrock and David P. Fleming		8. Performing Organization Report No. E-6099
9. Performing Organization Name and Address Lewis Research Center National Aeronautics and Space Administration Cleveland, Ohio 44135		10. Work Unit No. 129-03
		11. Contract or Grant No.
12. Sponsoring Agency Name and Address National Aeronautics and Space Administration Washington, D. C. 20546		13. Type of Report and Period Covered Technical Note
		14. Sponsoring Agency Code
15. Supplementary Notes		

16. Abstract

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17. Key Words (Suggested by Author(s)) Bearings, Self-acting bearings, Herringbone bearings, Optimization, Bearing load capacity	18. Distribution Statement Unclassified - unlimited

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price*
Unclassified	Unclassified	24	$3.00

*For sale by the National Technical Information Service, Springfield, Virginia 22151

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SUMMARY

A computer program was developed to determine the optimal herringbone groove parameters for maximum radial load capacity. The design curves shown in this report enable one to find the optimal herringbone journal bearing for a wide range of operating conditions. These include the following:

1. Incompressibly lubricated to highly compressible condition
2. Smooth or grooved member rotating
3. Length to diameter ratios of 1/4, 1/2, 1, and 2

The analysis is valid for small displacements of the journal center from the bearing center and for a large number of grooves.

Some of the findings of the work presented in this report are as follows:

1. For length to diameter ratios of 1 and 2 and small dimensionless bearing numbers, a plain journal bearing has a greater radial load capacity than any herringbone configuration. However, for the limiting case of incompressible lubrication there is a definite optimal herringbone configuration.
2. For dimensionless bearing number $\Lambda \rightarrow 0$, the incompressible case, the optimal configuration is the same whether the smooth or grooved member is rotating. However, as the bearing number increases, the optimal configuration differs appreciably depending on whether the smooth or groove member is rotating.
3. At high bearing numbers, the radial load capacity is appreciably higher for the case when the smooth member is rotating.

INTRODUCTION

More than any other factors, self-excited whirl instability and low load capacity limit the usefulness of gas lubricated self-acting journal bearings. The whirl problem is the tendency of the journal center to orbit the bearing center at an angular speed less than or equal to half that of the journal about its own center. In many cases the whirl amplitude is large enough to cause destructive contact of the bearing surfaces.

The low viscosity of gases results in low load capacity for self-acting gas lubricated journal bearings which is also a serious concern in many applications. Unlike liquid lubricants, a gaseous lubricant changes its density as it passes through the bearing. This so-called compressibility effect results in a "terminal" load condition. That is, the load capacity does not increase indefinitely with speed but quickly approaches a fixed value.

In quest of a bearing which would alleviate the two problems of self-excited whirl instability and low load capacity, Vohr and Chow (ref. 1) theoretically investigated a herringbone grooved journal bearing. They obtained a solution for bearing load capacity valid for small displacements of the journal center from the bearing center. An additional assumption was that the number of grooves was large enough that local pressure variations across a groove-ridge pair could be ignored. One of the conclusions obtained from the Vohr and Chow analysis is that, in contrast to a plain bearing, the load capacity of a herringbone-grooved journal bearing continues to increase without limit with increase in speed. Furthermore, the herringbone-grooved journal bearing may not suffer from the self-excited whirl instability that is normally associated with unloaded plain bearings. Malanoski (ref. 2) and Cunningham, Fleming, and Anderson (refs. 3 and 4) experimentally verified the aforementioned conclusion of Vohr and Chow.

Therefore, it has been shown that the self-acting herringbone journal bearing has highly desirable characteristics, namely that of high load capacity and that of operating in a whirl free condition. A remaining problem which is not in the literature is that of obtaining optimal herringbone journal bearing configurations for a wide range of bearing operating conditions. Therefore, the objective of the present report is to develop an optimization program, utilizing the analysis of Vohr and Chow (ref. 1) to determine groove configurations to maximize radial load capacity. Results are to be applicable for operating conditions ranging from an incompressible solution to a highly compressible solution ($\Lambda = 160$) and for bearing length to diameter ratios of 1/4 to 2.

SYMBOLS

b_1 width of groove

b_2 width of ridge

C dimensionless coefficient defined in appendix

D diameter of journal

e eccentricity of journal

F_r dimensionless radial load capacity of herringbone journal bearing, $f_r/\epsilon p_a LD$

$\overline{F}_r$ dimensionless radial load capacity of plain journal bearing, $\overline{f}_r/\epsilon p_a LD$

f_r radial load capacity of herringbone journal bearing

$\overline{f}_r$ radial load capacity of plain journal bearing

H_o film thickness ratio, h_{10}/h_{20}

$\overline{H}_o$ initial value of film thickness

h_{10} film thickness in groove region when journal is concentric

h_{20} film thickness in ridge region when journal is concentric

L length of journal

L_1 total axial length of groove

N number of grooves

p pressure

p_a ambient pressure

p_o zero order perturbation pressure or pressure when bearing is in concentric position

p_1 first-order perturbation pressure

R radius of journal

U velocity

W dimensionless load capacity of herringbone journal bearing, $w/\epsilon p_a LD$

W_D dimensionless load capacity of plain incompressibly lubricated journal bearing, $w_D/\epsilon p_a LD$

W_R dimensionless load capacity of plain compressibly lubricated journal bearing, $w_R/\epsilon p_a LD$

w total load capacity of herringbone journal bearing

w_D total load capacity of plain incompressibly lubricated journal bearing obtained from Donaldson (ref. 8)

w_R total load capacity of plain compressibly lubricated journal bearing obtained from Raimondi (ref. 9)

Z axial coordinate

α groove width ratio, $b_1/(b_1 + b_2)$

$\overline{\alpha}$ initial value of groove width ratio

β groove angle

$\overline{\beta}$ initial value of groove angle

γ groove length ratio, L_1/L

$\overline{\gamma}$ initial value of groove length ratio

Δ correction value

ϵ eccentricity ratio, e/h_{20}

θ angular coordinate

Λ dimensionless bearing number, $6\mu UR/p_a h_{20}^2$

λ length to diameter ratio, L/D

μ dynamic viscosity of fluid

ξ $\begin{cases} -1, \text{ grooved member rotating} \\ 1, \text{ smooth member rotating} \end{cases}$

BEARING DESCRIPTION

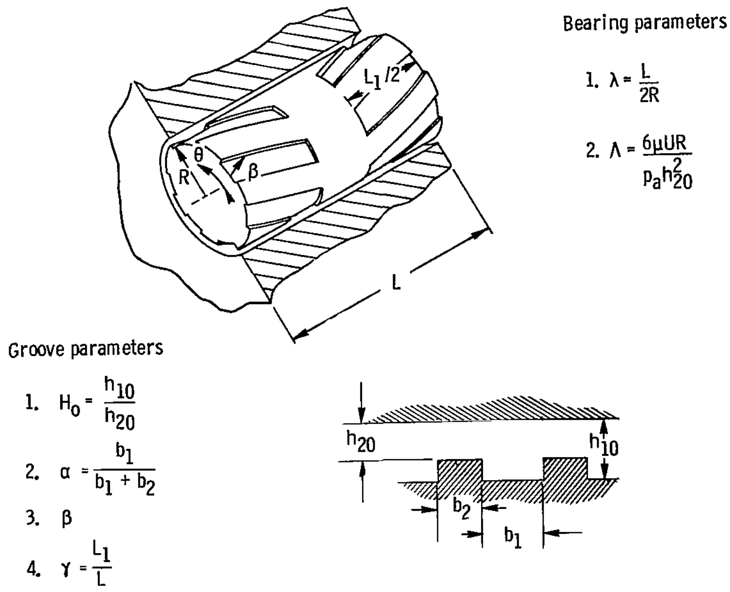
Sketch (a) shows the bearing to be studied. Note that the bearing has angled, shallow grooves in the journal surface. The grooves can be partial as shown or extend the complete length of the bearing. Also, the grooves can be placed in the rotating or nonrotating surfaces. The purpose of these grooves is to pump fluid toward the axial center of the bearing thereby increasing the lubricant pressure in the bearing. Load is directly related to the pressure distribution. This self-pressurization can increase the load capacity over that of a smooth bearing. The bearing shown in sketch (a) is unidirectional (i.e., it pumps inwardly for only one direction of rotation).

In sketch (a) the groove region is where the film thickness is h_{10} and the ridge is where the film thickness is h_{20}. Also the groove width is defined as b_1, and the ridge width is defined as b_2.

ANALYSIS

Equations for Herringbone-Grooved Bearing

Vohr and Chow (ref. 1), by assuming a large number of grooves, obtained rela-



(a)

tions for a "smoothed" pressure in the bearing film. That is, they dealt with an overall pressure rather than treating separately the pressure in the grooves and that over the ridges. They next assumed that the smoothed pressure $p(\theta, Z)$ could be represented by

$$p(\theta, Z) = p_o(Z) + \epsilon p_1(\theta, Z) \tag{1}$$

This is the well-known small eccentricity perturbation solution.

When equation (1) is substituted into the expressions for smoothed pressure, and terms collected according to the powers of ϵ, separate expressions result for p_o and p_1

$$\frac{d\,p_o}{d\,Z} = C_p \frac{p_a}{L} \tag{2}$$

$$C_1 \frac{L^2}{p_a} \frac{\partial^2 p_1}{\partial Z^2} + C_2 \frac{L}{p_a} \frac{\partial p_1}{\partial Z} + C_3 \frac{L}{p_a} \frac{\partial^2 p_1}{\partial Z\, \partial\theta} + \frac{C_4}{p_a} \frac{\partial p_1}{\partial\theta} + \frac{C_5}{p_a} \frac{\partial^2 p_1}{\partial\theta^2} + C_6 \sin\theta + C_7 \cos\theta = 0 \tag{3}$$

The coefficients C are given in the appendix. They differ slightly from the coefficients appearing in reference 1 because only steady conditions are considered here (no whirling) and because only one bearing member (grooved or smooth) is in motion. Once equations (1), (2), and (3) are solved for the pressure p, the bearing load may be calculated. These equations were derived for use with gas lubrication. However, they may also be used for incompressible lubricants by setting C_2, C_4, and C_7 to zero.

The coefficients C indicate that the groove parameters to be optimized are the following:

(1) The film thickness ratio H_o which is equal to the film thickness in the groove region divided by the film thickness in the ridge region when the bearing is concentric (i.e., $H_o = h_{10}/h_{20}$)

(2) The groove width ratio α which is equal to the width of the groove region divided by the width of the groove-ridge pair (i.e., $\alpha = b_1/(b_1 + b_2)$)

(3) The groove angle β

(4) The groove length ratio γ which is equal to the length covered by grooves divided by the overall length of the bearing (i.e., $\gamma = L_1/L$)

In sketch (a) the number of grooves is six. However, the analysis assumes essentially an infinite number of grooves. Reference 5 develops the following criterion for the minimum number of grooves such that the infinite groove analysis yields accurate results:

$$\frac{\Lambda}{N} < \frac{\left[(1 - \alpha)H_o^3 + \alpha\right]\left[H_o^3 + \alpha(1 - \alpha)(H_o^3 - 1)^2 \sin^2 \beta\right]}{2\pi\left[H_o^3 + \alpha(1 - \alpha)(H_o^3 - 1)^2\right](H_o - 1)\,\alpha(1 - \alpha)\sin^2 \beta} \tag{4}$$

where

N number of grooves

Λ dimensionless bearing number, $6\mu UR/p_a h_{20}^2$

The numerical value of the right side of (4) is typically between 5.5 and 8.0. Therefore, the minimum number of grooves placed around the journal can be represented conservatively by

$$N \geq \frac{\Lambda}{5} \tag{5}$$

Optimizing Procedure

The problem as defined in the INTRODUCTION is to find the optimal herringbone journal bearing for maximum radial load capacity for various bearing parameters. (The radial load capacity is the component in the direction of journal displacement of the total load capacity.) Therefore, the basic problem is to optimize the film thickness ratio H_o, the groove width ratio α, the groove angle β, and the groove length ratio γ for maximum radial load F_r given a dimensionless bearing number Λ, a length to diameter ratio λ, and whether the grooved ($\xi = -1$) or smooth ($\xi = 1$) member is rotating. Mathematically, this is expressed as follows:

Given: Λ, λ, and ξ.

Find: H_o, α, β, and γ which satisfy the following equation:

$$\frac{\partial F_r}{\partial H_o} = \frac{\partial F_r}{\partial \alpha} = \frac{\partial F_r}{\partial \beta} = \frac{\partial F_r}{\partial \gamma} = 0 \tag{6}$$

The method used is the Newton-Raphson method of solving simultaneous equations. This method is described in Scarborough (ref. 6); it was previously used in optimizing the step thrust bearing (ref. 7). Briefly, the method consists of letting

$$\left.\begin{aligned} H_o &= \overline{H}_o + \Delta H_o \\ \alpha &= \overline{\alpha} + \Delta\alpha \\ \beta &= \overline{\beta} + \Delta\beta \\ \gamma &= \overline{\gamma} + \Delta\gamma \end{aligned}\right\} \tag{7}$$

where $\overline{H}_o$, $\overline{\alpha}$, $\overline{\beta}$, and $\overline{\gamma}$ are initial values which might satisfy equation (6) and ΔH_o, $\Delta\alpha$, $\Delta\beta$, and $\Delta\gamma$ are correction terms. Substituting equations (7) into equation (6) and expanding these equations by Taylor's theorem for a function of four variables while neglecting second order and higher terms give the following:

$$\left.\begin{aligned}
\frac{\partial F_r}{\partial H_o} + \Delta H_o \frac{\partial^2 F_r}{\partial H_o^2} + \Delta\alpha \frac{\partial^2 F_r}{\partial H_o\, \partial\alpha} + \Delta\beta \frac{\partial^2 F_r}{\partial H_o\, \partial\beta} + \Delta\gamma \frac{\partial^2 F_r}{\partial H_o\, \partial\gamma} = 0 \\
\frac{\partial F_r}{\partial \alpha} + \Delta H_o \frac{\partial^2 F_r}{\partial\alpha\, \partial H_o} + \Delta\alpha \frac{\partial^2 F_r}{\partial\alpha^2} + \Delta\beta \frac{\partial^2 F_r}{\partial\alpha\, \partial\beta} + \Delta\gamma \frac{\partial^2 F_r}{\partial\alpha\, \partial\gamma} = 0 \\
\frac{\partial F_r}{\partial \beta} + \Delta H_o \frac{\partial^2 F_r}{\partial\beta\, \partial H_o} + \Delta\alpha \frac{\partial^2 F_r}{\partial\beta\, \partial\alpha} + \Delta\beta \frac{\partial^2 F_r}{\partial\beta^2} + \Delta\gamma \frac{\partial^2 F_r}{\partial\beta\, \partial\gamma} = 0 \\
\frac{\partial F_r}{\partial \gamma} + \Delta H_o \frac{\partial^2 F_r}{\partial\gamma\, \partial H_o} + \Delta\alpha \frac{\partial^2 F_r}{\partial\gamma\, \partial\alpha} + \Delta\beta \frac{\partial^2 F_r}{\partial\gamma\, \partial\beta} + \Delta\gamma \frac{\partial^2 F_r}{\partial\gamma^2} = 0
\end{aligned}\right\} \tag{8}$$

The partial derivatives in equations (8) can be expressed in terms of central difference formulation. The correction terms ΔH_o, $\Delta\alpha$, $\Delta\beta$, and $\Delta\gamma$ are found by using determinants. Additional corrections can be obtained by repeated application of equations (7) and (8) where the initial values $\overline{H}_o$, $\overline{\alpha}$, $\overline{\beta}$, and $\overline{\gamma}$ are now the values of H_o, α, β, and γ given by equation (7) of the preceding evaluation.

DISCUSSION OF RESULTS

A digital computer program was written to solve for the pressure distribution as expressed in equations (1), (2), and (3). Knowing the pressure, the load capacity was directly obtained. The dimensionless load capacity is a function of the groove parameters H_o, α, β, and γ, as well as the following bearing operating parameters:

(1) Dimensionless bearing number $\Lambda = 6\mu UR/p_a h_{20}^2$

(2) Length to diameter ratio $\lambda = L/D$

(3) Smooth member rotating ($\xi = 1$); grooved member rotating ($\xi = -1$)

Verification of Equations

Table I shows that the equations developed are indeed valid. The table compares the dimensionless load capacity of a plain journal with that of a herringbone journal bearing when it is made to approach a plain journal bearing. That is, table I shows the following:

$$\text{Herringbone journal when } \left\{ \begin{array}{c} H_0 \to 1 \\ \alpha \to 0 \\ \beta \to 0 \\ \gamma \to 0 \end{array} \right\} \to \text{Plain journal bearing}$$

Table I uses plain journal bearing results from Donaldson (ref. 8) for the incompressible results and Raimondi (ref. 9) for the compressible results. These results are for an eccentricity ratio of 0.1. From table I it is seen that the herringbone bearing loads, when $H_0 \to 1$ and $\alpha \to \beta \to \gamma \to 0$, are within 3 percent of the plain journal bearing results.

Table I also shows that, for incompressible lubrication, the dimensionless load decreases 85 percent as the length to diameter ratio changes from 2.0 to 0.5. For the compressible case of $\Lambda = 6$, the dimensionless load decreases only 53 percent. This implies that side leakage increases at a much greater rate for incompressible lubrication.

Optimization Results

Tables II and III give optimal herringbone parameters (H_0, α, β, γ) for maximum radial load capacity F_r. The difference between these tables is that in table II the grooved member is rotating and in table III the smooth member is rotating. Tables II and III cover a wide range of bearing parameters: from incompressible lubrication ($\Lambda \to 0$) to a highly compressible situation ($\Lambda = 160$) and length to diameter ratios of 1/4, 1/2, 1, and 2. Tables II and III also contain the radial load capacity of a plain journal bearing $\overline{F}_r$.

The following observations can be made from tables II and III:

(1) For a length to diameter ratio λ of 2 and dimensionless bearing number Λ of 1 the optimal configuration is a plain journal bearing.

(2) At high values of dimensionless bearing numbers Λ, the radial load capacity of an optimal herringbone journal bearing is considerably higher than that of a plain journal bearing.

(3) For incompressible lubrication, the results are exactly the same whether the smooth or grooved member is rotating.

(4) As the dimensionless bearing number Λ increases, the optimal configuration

differs depending on whether the smooth or grooved member is rotating.

(5) For $\lambda = 1/4$ there is little change in optimal configuration over the complete range of bearing number. However, for $\lambda = 2$ there is considerable change in the optimal configuration.

Figures 1 to 5 are directly obtained from data presented in tables II and III. In the top graph of each figure, the grooved member is assumed to be rotating; in the bottom graph, the smooth member is assumed to be rotating. Figures 1 to 4 show the effect of Λ on optimal configuration parameters for $\lambda = 1/4, 1/2, 1$, and 2. In all these figures it is observed that, for $\lambda = 1$ and $\lambda = 2$ and small dimensionless bearing numbers, the results change rapidly and tend toward a plain bearing as optimal. However, it should be pointed out that, for the limiting case of incompressible lubrication, the plain bearing is not optimal but the herringbone configuration is as shown in tables II and III. For some cases in the low dimensionless bearing number range ($\Lambda < 24$), additional data other than that shown in tables II and III had to be obtained to produce the curves of figures 1 to 5.

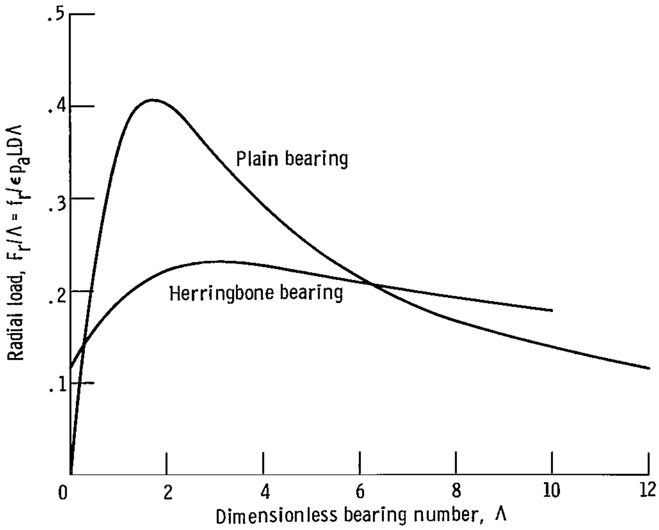
Therefore, given the bearing operating conditions (Λ, λ, and ξ) and using figures 1 to 5 one can easily obtain the optimal herringbone bearing configuration.

Resulting Radial Load

Figure 5 shows the effect of Λ on dimensionless radial load capacity of an optimal herringbone journal bearing for length to diameter ratios of 1/4, 1/2, 1, and 2. It is observed that the dimensionless radial load capacity increases with Λ and does not approach any fixed value as is the case for a plain journal bearing. It is seen from figure 5 that the dimensionless radial load capacity is appreciably higher for the case when the smooth member is rotating. Also shown in figure 5 are the plain bearing results for $\lambda = 2$. Note that the herringbone bearing does not result in a "terminal" load condition as is true for the plain journal bearing.

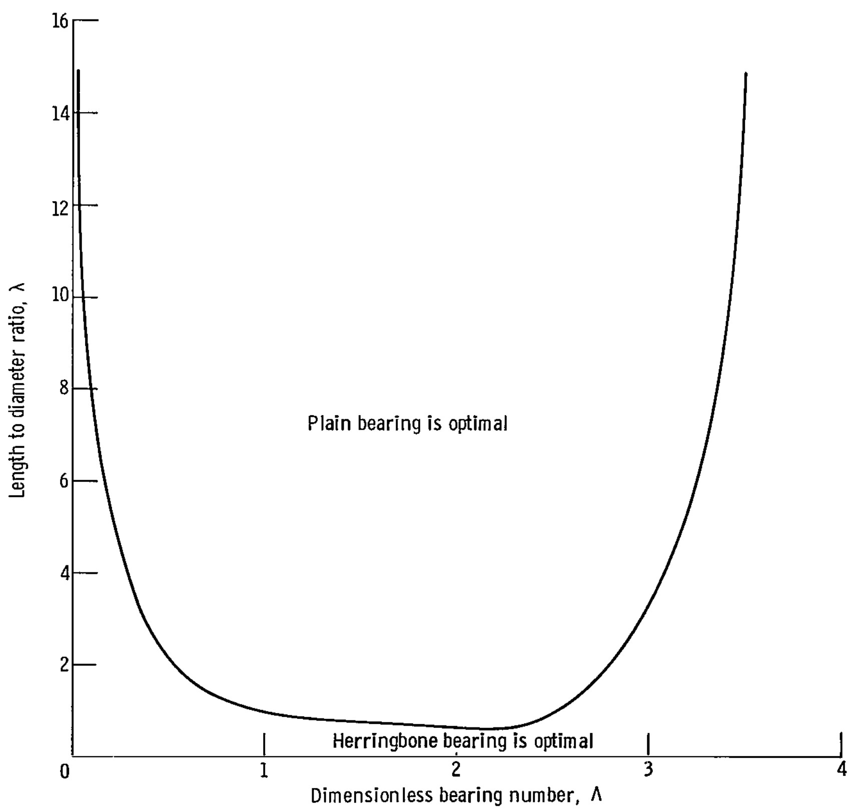
EXPLANATION OF RESULTS

The reason why, under certain bearing conditions, the plain bearing is optimal is shown in sketch (b). When the bearing number $\Lambda = 0$, $F_r/\Lambda = 0$ for a plain bearing, whereas $F_r/\Lambda > 0$ for a herringbone bearing. The radial load capacity of a plain bearing increases with Λ much faster in a plain bearing, however, and rises above F_r for a herringbone bearing. Since the radial load capacity of a herringbone bearing increases indefinitely with Λ, while that for a plain bearing approaches a fixed limit, the herring-



.5
.4
.3
.2
.1
Radial load, $F_r/\Lambda = f_r/\epsilon p_a LD\Lambda$
Plain bearing
Herringbone bearing
0
2
4
6
8
10
12
Dimensionless bearing number, Λ

(b)



16
14
12
10
8
6
4
2
Length to diameter ratio, λ
Plain bearing is optimal
Herringbone bearing is optimal
0
1
2
3
4
Dimensionless bearing number, Λ

(c)

bone bearing has the higher load capacity at high Λ. Sketch (c), which plots bearing number Λ against length to diameter ratio λ, shows the region where the plain bearing yields a higher radial load capacity than a herringbone bearing (smooth member is rotating).

CONCLUDING REMARKS

A computer program was developed to determine optimal self-acting herringbone journal bearings for maximum radial load capacity. The design curves shown in the report enable one to find the optimal herringbone journal bearing for a wide range of operating conditions. These range from incompressible lubrication to a highly compressible condition, for either smooth or grooved members rotating, and for length to diameter ratios of 1/4, 1/2, 1, and 2. The analysis is valid for small displacement of the journal center from the bearing center and for a large number of grooves.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, February 25, 1971,
129-03.

APPENDIX - COEFFICIENTS FOR HERRINGBONE BEARING EQUATIONS

$$C_p = \frac{\alpha(1 - \alpha)(H_o^3 - 1)(H_o - 1) \sin\beta \cos\beta}{H_o^3 + \alpha(1 - \alpha)(H_o^3 - 1)^2 \sin^2\beta} 2\Lambda\lambda$$

$$C_1 = \frac{H_o^3 + \alpha(1 - \alpha)(H_o^3 - 1)^2 \sin^2\beta}{\alpha + (1 - \alpha)H_o^3} \frac{1}{2\lambda} \frac{p_o}{p_a}$$

$$C_2 = \frac{\alpha(1 - \alpha)(H_o^3 - 1)(H_o - 1) \sin\beta \cos\beta}{\alpha + (1 - \alpha)H_o^3} \Lambda$$

$$C_3 = \frac{2\xi\alpha(1 - \alpha)(H_o^3 - 1)^2 \sin\beta \cos\beta}{\alpha + (1 - \alpha)H_o^3} \frac{p_o}{p_a}$$

$$C_4 = C_3 C_p \frac{p_a}{p_o} + \frac{\alpha(1 - \alpha)(H_o^3 - 1)(H_o - 1) \sin^2\beta}{\alpha + (1 - \alpha)H_o^3} 2\xi\lambda\Lambda - 2\lambda\Lambda(1 - \alpha + \alpha H_o)$$

$$C_5 = \frac{H_o^3 + \alpha(1 - \alpha)(H_o^3 - 1)^2 \cos^2\beta}{\alpha + (1 - \alpha)H_o^3} 2\lambda \frac{p_o}{p_a}$$

$$C_6 = \left\{ \frac{-3\xi\alpha(1 - \alpha)(H_o - 1)^2 \sin^2\beta}{\left[\alpha + (1 - \alpha)H_o^3\right]^2} \cdot \left[\frac{\alpha(1 - \alpha)(H_o^3 - 1)^2 \cos^2\beta \left[(1 - \alpha)H_o^2(H_o^2 + H_o - 1) - \alpha(H_o^2 - H_o - 1)\right]}{H_o^3 + \alpha(1 - \alpha)(H_o^3 - 1)^2 \sin^2\beta} - H_o^2 \right] + 1 \right\}$$

$$C_7 = \frac{3C_p^2(H_o^2 - 1)}{\left[\alpha + (1 - \alpha)H_o^3\right](H_o^3 - 1)} \left[H_o^2(H_o^2 + 1) + \alpha(1 - \alpha)(H_o^3 - 1)^2 \sin^2 \beta\right] \frac{1}{2\lambda}$$

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TABLE I. - COMPARISON OF DIMENSIONLESS LOAD CAPACITY OF PLAIN JOURNAL BEARING WITH THAT OF HERRINGBONE JOURNAL BEARING AS IT APPROACHES PLAIN JOURNAL BEARING

[Eccentricity ratio, 0.1.]

	Incompressible lubrication	Compressible lubrication							Length to diameter ratio, λ
		Dimensionless bearing number, Λ							
		1.24	0.6	1.2	3	6	12	24	
Dimensionless load capacity of herringbone journal bearing, W	------	0.0286	0.0713	0.142	0.344	0.625	0.961	1.19	0.5
Dimensionless load capacity of plain journal bearing (from ref. 9), W_R	------	.0288	.0719	.143	.348	.634	.974	1.22	
W/Λ	0.1190	------	------	-----	-----	-----	-----	----	
W_D/Λ	.1205	------	------	-----	-----	-----	-----	----	
Dimensionless load capacity of herringbone journal bearing, W	------	0.0897	0.221	0.425	0.851	1.13	1.28	----	1
Dimensionless load capacity of plain journal bearing (from ref. 9), W_R	------	.0902	.223	.429	.862	1.16	1.31	----	
W/Λ	0.3745	------	------	-----	-----	-----	-----	----	
W_D/Λ	.3776	------	------	-----	-----	-----	-----	----	
Dimensionless load capacity of herringbone journal bearing, W	------	0.193	0.458	0.786	1.18	1.33	-----	----	2
Dimensionless load capacity of plain journal bearing (from ref. 9), W_R	------	.194	.462	.794	1.20	1.36	-----	----	
W/Λ	0.8137	------	------	-----	-----	-----	-----	----	
W_D/Λ	.8170	------	------	-----	-----	-----	-----	----	

TABLE II. - OPTIMAL HERRINGBONE GROOVE PARAMETERS FOR MAXIMUM DIMENSIONLESS RADIAL LOAD CAPACITY FOR VARIOUS DIMENSIONLESS BEARING NUMBERS AND LENGTH TO DIAMETER RATIOS WHEN GROOVED MEMBER IS ROTATING

	Incompressible solution	Compressible lubrication							Length to diameter ratio, λ
		Dimensionless bearing number, Λ							
		0.1	1	10	20	40	80	160	
Film thickness ratio, H_o	2.592	2.592	2.588	2.548	2.507	2.471	2.489	2.552	
Groove width ratio, α	.5006	.5002	.4973	.4773	.4703	.4823	.5234	.5604	
Groove length ratio, γ	.9416	.9410	.9408	.9201	.8779	.7961	.7015	.6286	
Groove angle, β, deg	19.26	19.26	19.25	19.40	19.88	20.91	22.31	23.69	
Dimensionless radial load capacity of herringbone journal bearing, F_r	-------	.0038	.0381	.3864	.7579	1.396	2.337	3.625	$\frac{1}{4}$
Dimensionless radial load capacity of plain journal bearing, $\overline{F}_r$	-------	0	.0008	.0734	.2503	.631	1.021	1.222	
F_r/Λ	.0380	-------	-------	-------	-------	-------	-------	-------	
$\overline{F}_r/\Lambda$	0	-------	-------	-------	-------	-------	-------	-------	
Film thickness ratio, H_o	2.381	2.379	2.355	2.241	2.317	2.454	2.567	2.617	
Groove width ratio, α	.5044	.5021	.4820	.3577	.3799	.4416	.4552	.4253	
Groove length ratio, γ	.8653	.8653	.8636	.7463	.6247	.5483	.5146	.5327	
Groove angle, β, deg	23.26	23.25	23.21	22.82	22.68	23.31	23.88	23.83	
Dimensionless radial load capacity of herringbone journal bearing, F_r	-------	.0067	.0689	.7362	1.295	1.985	2.913	4.482	$\frac{1}{2}$
Dimensionless radial load capacity of plain journal bearing, $\overline{F}_r$	-------	.0001	.0107	.5855	.990	1.211	1.318	1.393	
F_r/Λ	.0669	-------	-------	-------	-------	-------	-------	-------	
$\overline{F}_r/\Lambda$	0	-------	-------	-------	-------	-------	-------	-------	
Film thickness ratio, H_o	2.219	2.203	2.051	2.294	2.486	2.587	2.616	2.630	
Groove width ratio, α	.5228	.5089	.3529	.2951	.3816	.3913	.3725	.3632	
Groove length ratio, γ	.7607	.7567	.6857	.4213	.4397	.4785	.5503	.6032	
Groove angle, β, deg	28.62	28.59	27.94	23.38	23.77	23.71	23.18	22.62	
Dimensionless radial load capacity of herringbone journal bearing, F_r	-------	.0103	.1234	1.334	1.876	2.691	4.229	7.367	1
Dimensionless radial load capacity of plain journal bearing, $\overline{F}_r$	-------	.0011	.0980	1.160	1.302	1.388	1.444	1.482	
F_r/Λ	.1007	-------	-------	-------	-------	-------	-------	-------	
$\overline{F}_r/\Lambda$	0	-------	-------	-------	-------	-------	-------	-------	
Film thickness ratio, H_o	2.147	2.085	1.0	2.404	2.503	2.566	2.624	2.661	
Groove width ratio, α	.5671	.5071	0	.3531	.3571	.3548	.3626	.3704	
Groove length ratio, γ	.6796	.6241	0	.3954	.4974	.5790	.6108	.6213	
Groove angle, β, deg	35.36	34.63	0	25.81	24.60	23.46	22.73	22.33	
Dimensionless radial load capacity of herringbone journal bearing, F_r	-------	.0126	.3557	1.792	2.580	4.172	7.417	13.95	2
Dimensionless radial load capacity of plain journal bearing, $\overline{F}_r$	-------	.0049	.3557	1.357	1.434	1.479	1.507	1.53	
F_r/Λ	.1138	-------	-------	-------	-------	-------	-------	-------	
$\overline{F}_r/\Lambda$	0	-------	-------	-------	-------	-------	-------	-------	

TABLE III. - OPTIMAL HERRINGBONE GROOVE PARAMETERS FOR MAXIMUM DIMENSIONLESS RADIAL LOAD CAPACITY FOR VARIOUS DIMENSIONLESS BEARING NUMBERS AND LENGTH TO DIAMETER RATIOS WHEN SMOOTH MEMBER IS ROTATING

	Incompressible solution	Compressible lubrication							Length to diameter ratio, λ
		Dimensionless bearing number, Λ							
		0.1	1	10	20	40	80	160	
Film thickness ratio, H_0	2.592	2.592	2.589	2.561	2.539	2.537	2.605	2.747	
Groove width ratio, α	.5006	.5003	.4979	.4804	.4739	.4859	.5246	.5640	
Groove length ratio, γ	.9416	.9411	.9420	.9443	.9360	.9096	.8766	.8591	
Groove angle, β, deg	19.26	19.26	19.24	19.26	19.65	20.79	22.42	23.73	
Dimensionless radial load capacity of herringbone journal bearing, F_r	-------	.0038	.0381	.3889	.7777	1.496	2.693	4.621	$\frac{1}{4}$
Dimensionless radial load capacity of plain journal bearing, $\overline{F}_r$	-------	0	.0008	.0734	.2503	.6309	1.021	1.222	
F_r/Λ	.0380	-------	-------	-------	-------	-------	-------	-------	
$\overline{F}_r/\Lambda$	0	-------	-------	-------	-------	-------	-------	-------	
Film thickness ratio, H_0	2.381	2.378	2.350	2.205	2.288	2.485	2.721	2.950	
Groove width ratio, α	.5044	.5025	.4854	.3810	.4236	.4967	.5327	.5326	
Groove length ratio, γ	.8653	.8655	.8674	.8480	.8166	.8146	.8328	.8575	
Groove angle, β, deg	23.26	23.25	23.23	24.11	25.47	26.64	26.75	25.60	
Dimensionless radial load capacity of herringbone journal bearing, F_r	-------	.0067	.0686	.7544	1.390	2.319	3.792	6.438	$\frac{1}{2}$
Dimensionless radial load capacity of plain journal bearing, $\overline{F}_r$	-------	.0001	.0107	.5855	.9900	1.211	1.318	1.393	
F_r/Λ	.0669	-------	-------	-------	-------	-------	-------	-------	
$\overline{F}_r/\Lambda$	0	-------	-------	-------	-------	-------	-------	-------	
Film thickness ratio, H_0	2.219	2.201	2.022	2.113	2.338	2.557	2.723	2.828	
Groove width ratio, α	.5228	.5112	.3570	.3013	.4104	.4521	.4643	.4695	
Groove length ratio, γ	.7607	.7564	.6773	.5754	.6897	.7678	.8113	.8330	
Groove angle, β, deg	28.62	28.58	27.81	28.70	30.27	29.69	28.36	27.36	
Dimensionless radial load capacity of herringbone journal bearing, F_r	-------	.0102	.1212	1.363	2.061	3.264	5.567	10.15	1
Dimensionless radial load capacity of plain journal bearing, $\overline{F}_r$	-------	.0011	.0980	1.160	1.302	1.388	1.444	1.482	
F_r/Λ	.1007	-------	-------	-------	-------	-------	-------	-------	
$\overline{F}_r/\Lambda$	0	-------	-------	-------	-------	-------	-------	-------	
Film thickness ratio, H_0	2.147	2.077	1.0	2.218	2.389	2.501	2.565	2.602	
Groove width ratio, α	.5671	.5159	0	.3626	.4015	.4157	.4221	.4261	
Groove length ratio, γ	.6796	.6179	0	.5492	.6287	.6765	.7028	.7151	
Groove angle, β, deg	35.36	34.45	0	32.84	31.62	30.31	29.44	28.90	
Dimensionless radial load capacity of herringbone journal bearing, F_r	-------	.0125	.3557	1.906	2.946	4.976	9.020	17.11	2
Dimensionless radial load capacity of plain journal bearing, $\overline{F}_r$	-------	.0049	.3557	1.357	1.434	1.479	1.507	1.526	
F_r/Λ	.1138	-------	-------	-------	-------	-------	-------	-------	
$\overline{F}_r/\Lambda$	0	-------	-------	-------	-------	-------	-------	-------	

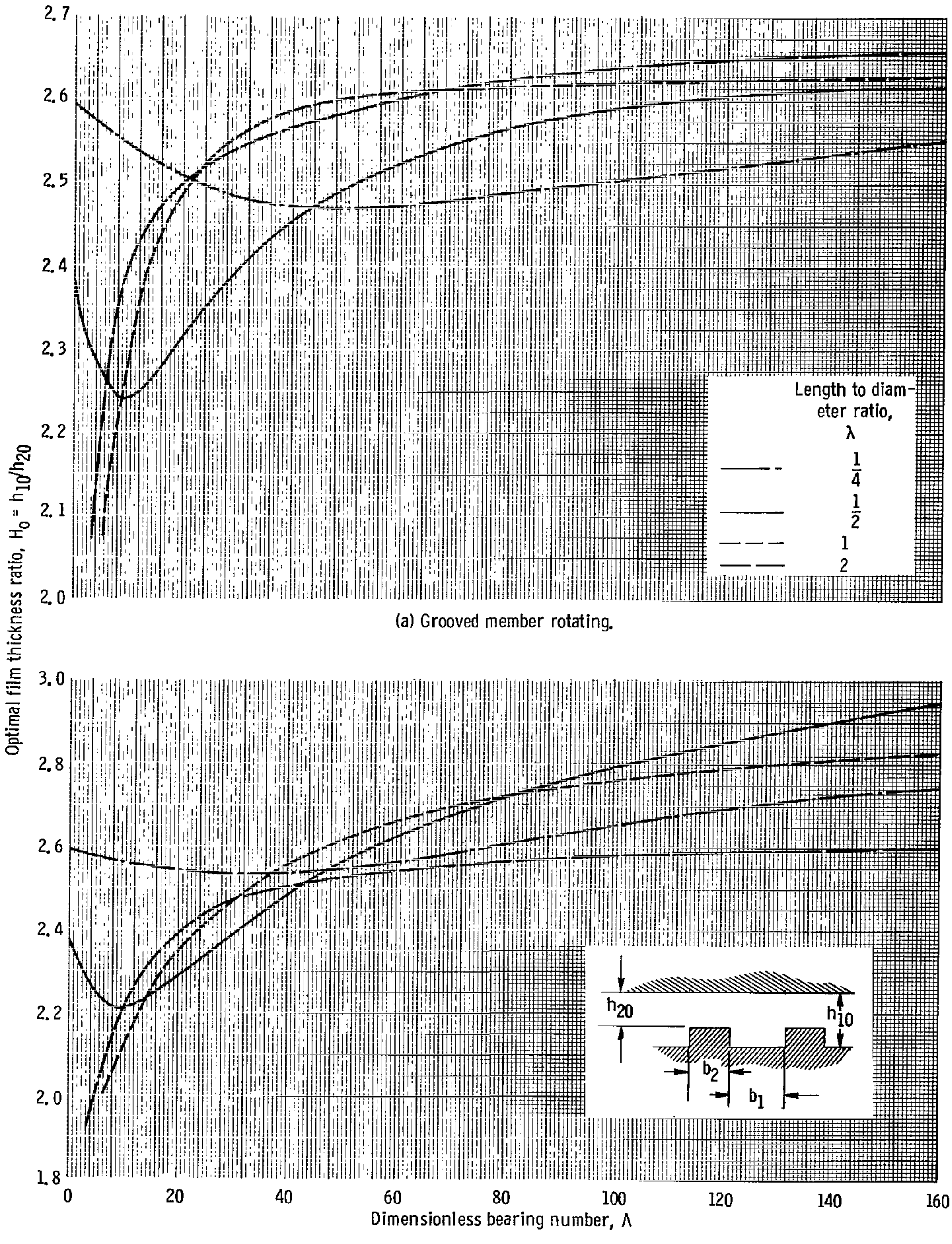
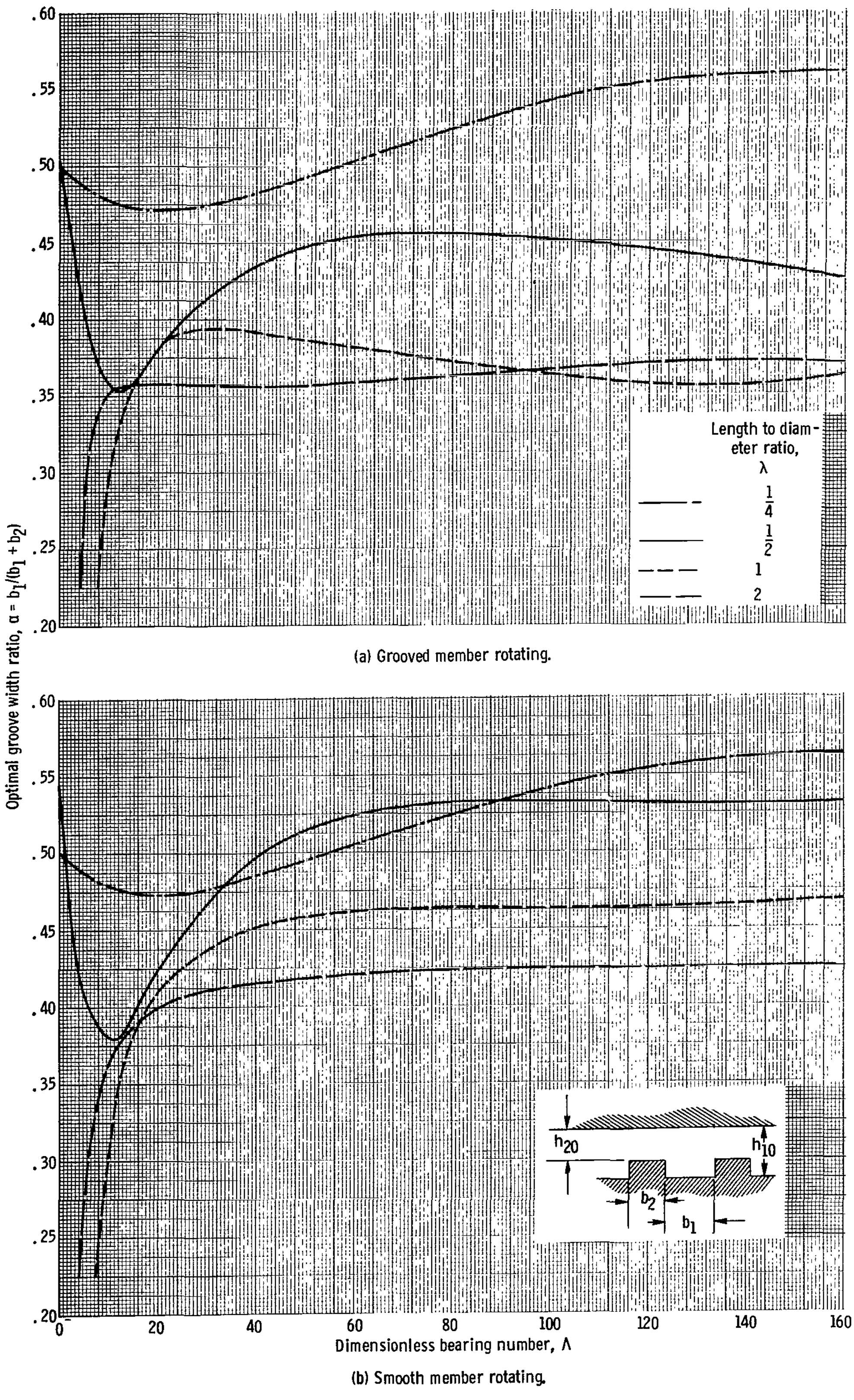


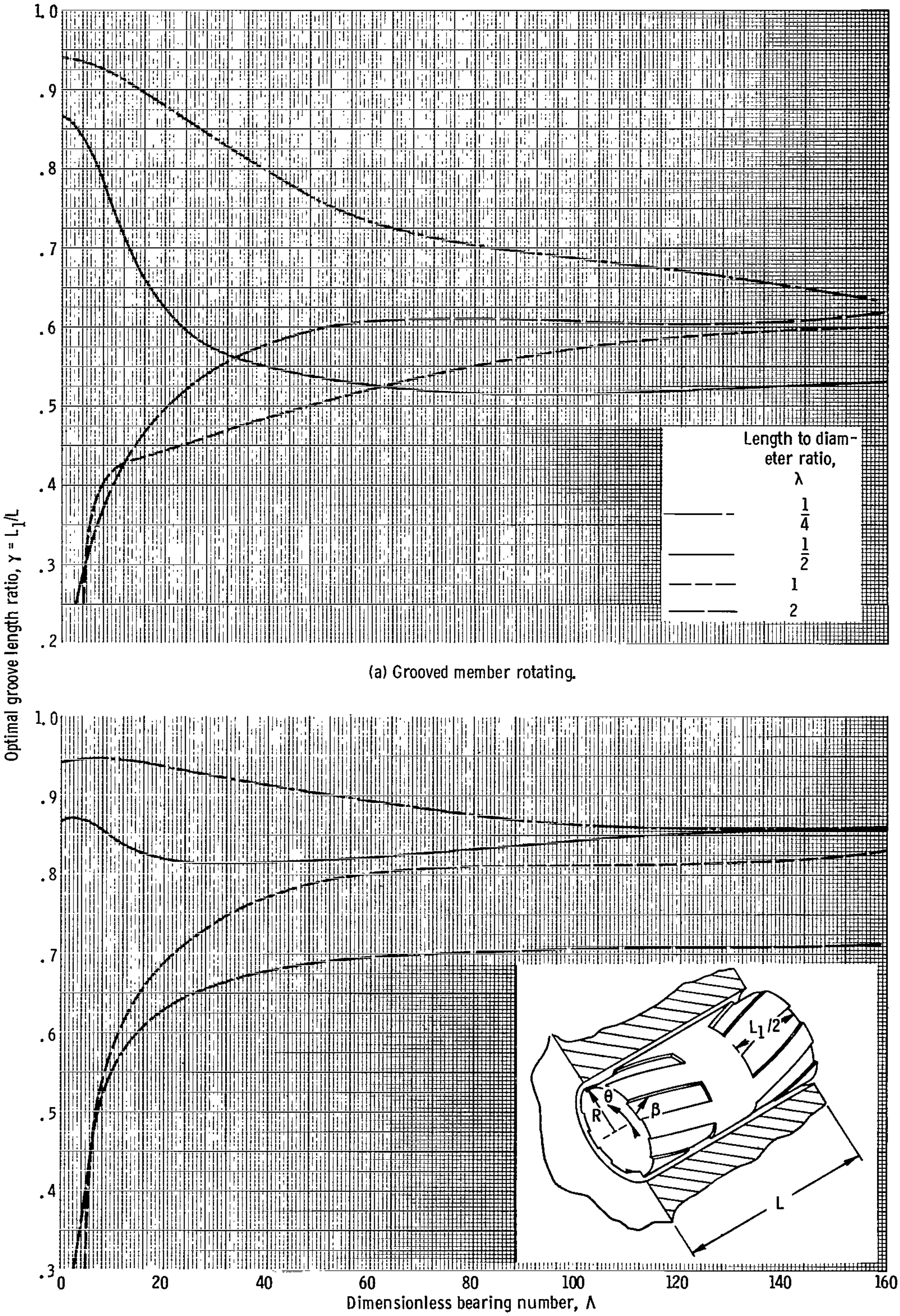
Figure 1. - Film thickness ratio to maximize radial load capacity of herringbone groove bearing.



(a) Grooved member rotating.

(b) Smooth member rotating.

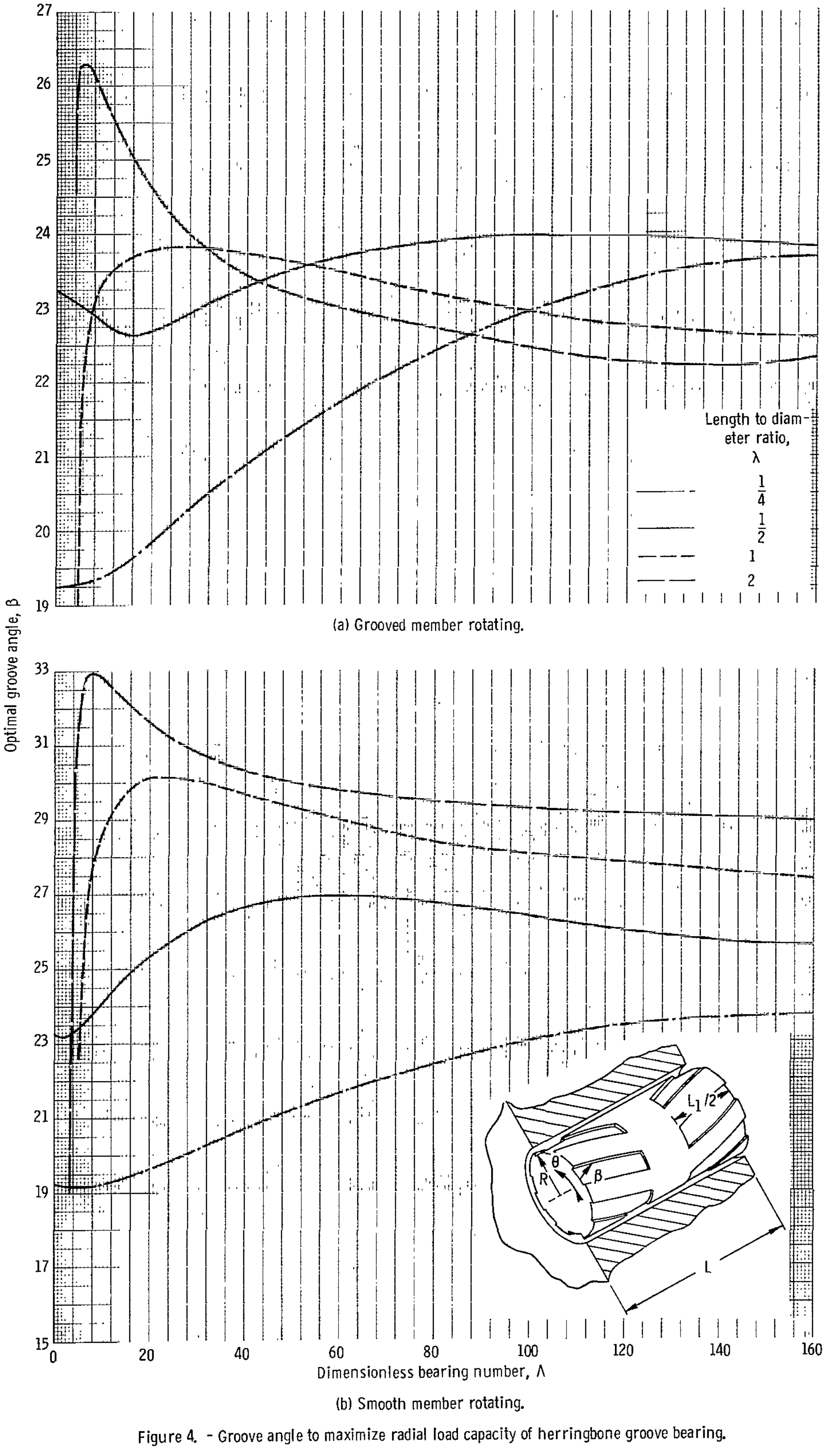
Figure 2. - Groove width ratio to maximize radial load capacity of herringbone groove bearing.



(a) Grooved member rotating.

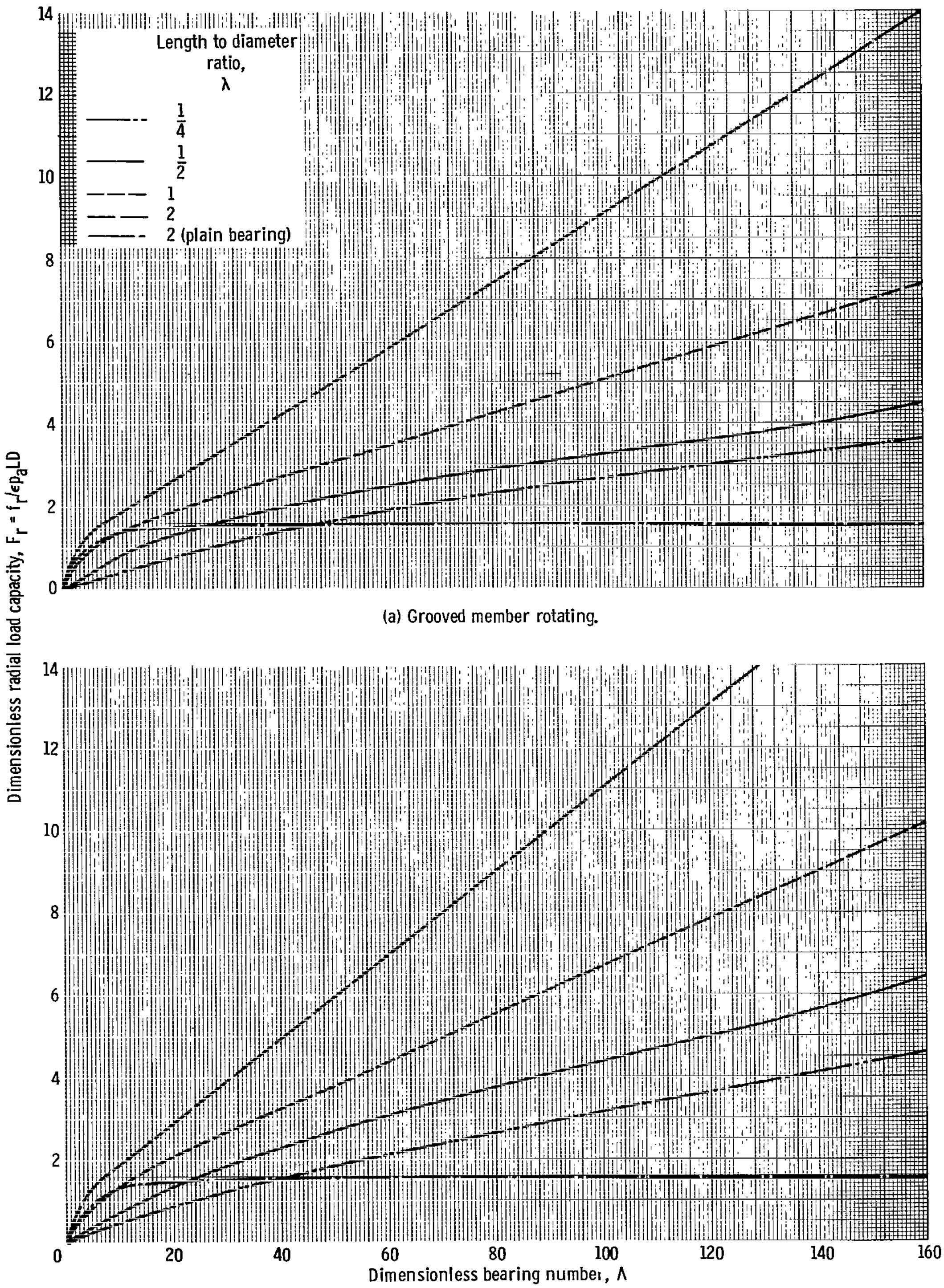
(b) Smooth member rotating.

Figure 3. - Groove length ratio to maximize radial load capacity of herringbone groove bearing.



(b) Smooth member rotating.

Figure 4. - Groove angle to maximize radial load capacity of herringbone groove bearing.



(a) Grooved member rotating.

(b) Smooth member rotating.

Figure 5. - Radial load capacity of optimal herringbone groove bearing.

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